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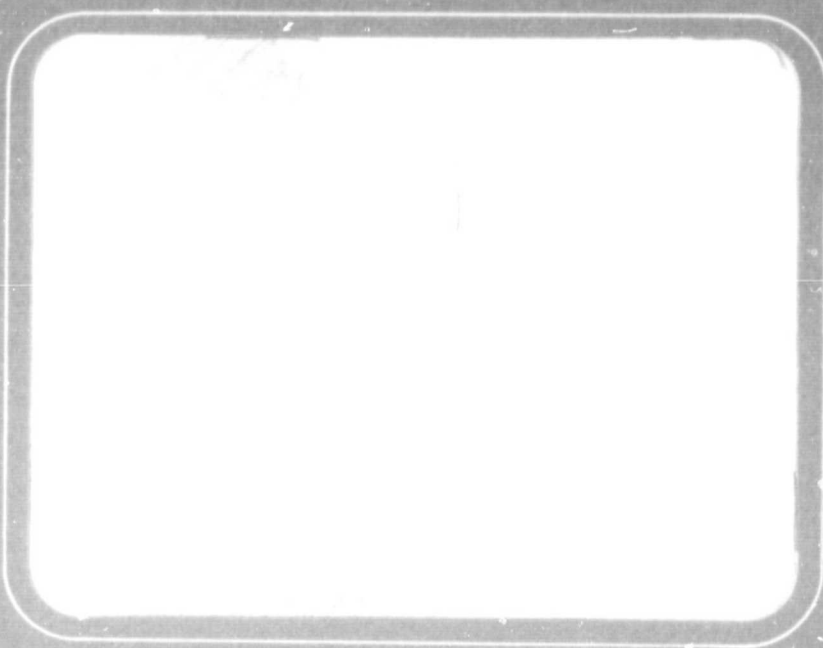
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Report



FINAL REPORT

on

**EVALUATION OF BEARING MOUNTING DESIGN
AND EXCESSIVE WEAR PHENOMENA
(Contract NAS8-33576, Task No. 108)**

to

**NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
GEORGE C. MARSHALL SPACE FLIGHT CENTER**

June 21, 1982

by

J. W. Kannel

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INTRODUCTION

Efforts to improve the performance life of the space shuttle main engine (SSME) bearing are continuing. A reusable design lifetime of 7.5 hours is being sought which requires a significant extension of bearing technology for this type of application. The bearings for the high pressure oxygen turbopump (HPO[™]) are of particular concern because of the high DN* (1.7×10^6) and the fact that they must operate with very poor lubrication. Most of the lubrication is achieved by solid transfer of the cage material (PTFE) to the races.

Some improvement in bearing performance has been achieved through improvement in the ball-race lubricant transfer. In our last study,^{(1)**} techniques for enhancing transfer were evaluated including precoating the balls with molybdenum disulfide (a suggestion evolved from the Hughes-DARPA solid film research effort).⁽²⁾ Work at Rocketdyne in this area of lubrication appears to offer some very good promise for improving bearing performance.

A second area for improvement in the bearings at Rocketdyne has been to modify the bearing preload springs.⁽³⁾ Four preload spring

*Design Number (DN) = Bore diameter (mm) x rpm

**References are listed on page 18.

arrangements have been evaluated as follows:

- (1) Belleville spring
- (2) Short-short beam spring
- (3) Short-tall beam spring
- (4) Modified short-tall beam spring.

Each of these spring arrangements have been designed to improve the pre-load consistency of the HPOTP bearings; however, life limitations are still occurring.

The purpose of this current task has been to evaluate the effect of bearing thermal growth on the effectiveness of this spring. The specific objectives of this task have been as follows:

- (1) Evaluate the SSME HPOTP turbine end bearing, preload spring, and bearing mounting design relative to the current bearing problems, i.e.,
 - (a) spalling
 - (b) excessive ball wear
 - (c) possible thermal problem
 - (d) cage delamination.
- (2) Assume the excessive, uniform ball wear phenomenon to be the result of a thermal problem and that ball wear of 0.076, 0.152 and 0.279 mm (0.003, 0.006, and 0.001 inch) is caused by excessive thermal stress. Calculate the magnitude of the thermal problem to cause these levels of ball wear.
- (3) Create plots of maximum shear stress and maximum reversing shear stress versus axial load for the 57-mm (2.24-inch) SSME HPOTP bearing.
- (4) Create a plot of bearing thermal growth versus preload spring deflection.

SUMMARY AND CONCLUSIONS

(1) Metallic wear can easily be a source of enlargement of the contact zone between ball and races. This wear could occur as a result of poor lubrication and could cause a movement in the preload spring. Thermal growth in the bearing could contribute to this wear but probably not extensively.

(2) As the loads are increased in a bearing, very high fatigue-inducing shear stresses will be generated. The depth of these stresses could easily be on the order of 0.1 to 0.2mm (0.004 to 0.008 inch). Further, since fatigue occurs in a region rather than at a fixed depth, the failure depth could be as much as 0.28mm (0.011 inch) for a 35,580N (8000 pound) load.

(3) The preload spring designs appear to operate satisfactorily for a temperature differential of around 100C (210F). At temperatures in the 100 to 150C (210 to 300F) range, however, the springs bottom out and very high loads are developed in the bearing. It must be noted that the preload springs allow very little room for fabrication error or tolerance stack-up. It appears that more concern should be given to allowable spring movement after assembly.

PROJECT DETAILS

Discussion of Bearing Failure Modes

The purpose of the study was to evaluate bearing distress as a result of a malfunction of the preloading arrangement for the bearings. Before discussing the specific type of analyses developed in the project, it is important to discuss possible bearing failure modes qualitatively.

Spalling

Spalling of a bearing is normally associated with a fatigue failure. Here, a spall is formed as a result of an impurity or inclusion beneath the surface of the balls or races. Repeated cycling of the

stresses around this inclusion cause a fatigue crack to be initiated. Eventually, this crack grows to the surface and a spall is formed.

The cyclic stresses normally considered to initiate a fatigue spall are the reversing shear stresses or the maximum shear stresses that occur as a result of the ball rolling on the races. These stresses vary beneath the surface and reach a peak at a depth of .25 times the contact width in the rolling direction. For the SSME bearing, this is on the order of 0.076 to 0.127mm (.002 to .005 inch) beneath the surface.

Spall formation rate is a strong function of the bearing loading and any increase in load will cause a marked drop in the fatigue life. For this reason, the bearing spring load should be held as near the design level as possible. If, for example, the bearing load is increased by 50 percent, theoretically, the life will decrease by a factor of 3 to 4.

Excessive Ball Wear

Excessive ball wear presumably occurs as a result of metallic contact between the balls and races. This action is heavily affected by the presence (or absence) of a "third-body" between balls and races. In liquid lubrication, the third body is a thin film of lubricant. With transfer film lubrication, this third body is a solid layer (such as PTFE) transferred from the cage. Ideally, the third body becomes the sacrificial layer to protect the balls and races. If the layer is not continuously replenished, metal wear will occur and will eventually destroy the bearing.

The general observation from the Battelle evaluation is that lubrication is the key to SSME bearing success. Other factors such as preloading or thermal problems can be minimized with good lubrication.

Possible Thermal Problems

The SSME is literally a plethora of heat sources and sinks. That is, the pumped fluid is at cryogenic temperature and passes through a bearing where it may absorb 15-22kw (20-30 HP). The resultant temperature gradient can result in large shaft housing expansions and contractions and cause bearing overloads. One effect is the growth of and

eventual lockup of the bearing cartridge.⁽⁴⁾ Figure 1 shows the effect of bearing load and ball-cage friction on thermal problems. If the friction coefficient is greater than 0.1, cartridge lockup is probable. This implies that if the bearing does not have good lubrication, cartridge lockup and subsequent bearing failure is a strong possibility.

Cage Delamination

Cage delamination problems are probably the result of thermal stresses coupled with inadequate cage fabrication. Without specific failure analyses and laboratory testing, the extent of seriousness of this problem is difficult to assess. Under severe service circumstances, improper curing of the cage materials can lead to cage breakage and bearing failure. However, it is difficult to assume that minor delamination will lead to severe failure.

Effect of Thermal Growth on Bearing Wear

The objective of this analyses was to evaluate the effect of thermal growth of the bearing on wear depth. One evaluation involved estimating the increase in the contact region as a function of ball-race conformal wear, Figure 2. Figure 3 shows the result of these estimations. To illustrate what the computations show, consider a case having a ball-race contact width of 1.5mm (0.060 inch). Suppose as a result of wear that a wear scar is generated on the race with a half contact width of 1.27mm (0.05 inch). The ratio of scar width to contact width is 0.822. From Figure 3, actual contact width for this same load is 1.15 times the original contact width, or 1.75mm (0.07 inch). That is, as wear occurs the contact region grows in proportion to the wear. If, the wear mechanism is not impeded (such as by minimal lubrication), the entire raceway will eventually be destroyed.

As the ball-raceway interface wears, the bearing will move axially as a result of the preload spring. Figure 4 presents an estimate of the spring movement (these calculations were based simply on the sine of the contact angle). As a example, the preload spring would move about 0.254mm (.010 inch) for a wear of 0.132mm (.0052 inch), which is not unreasonable.

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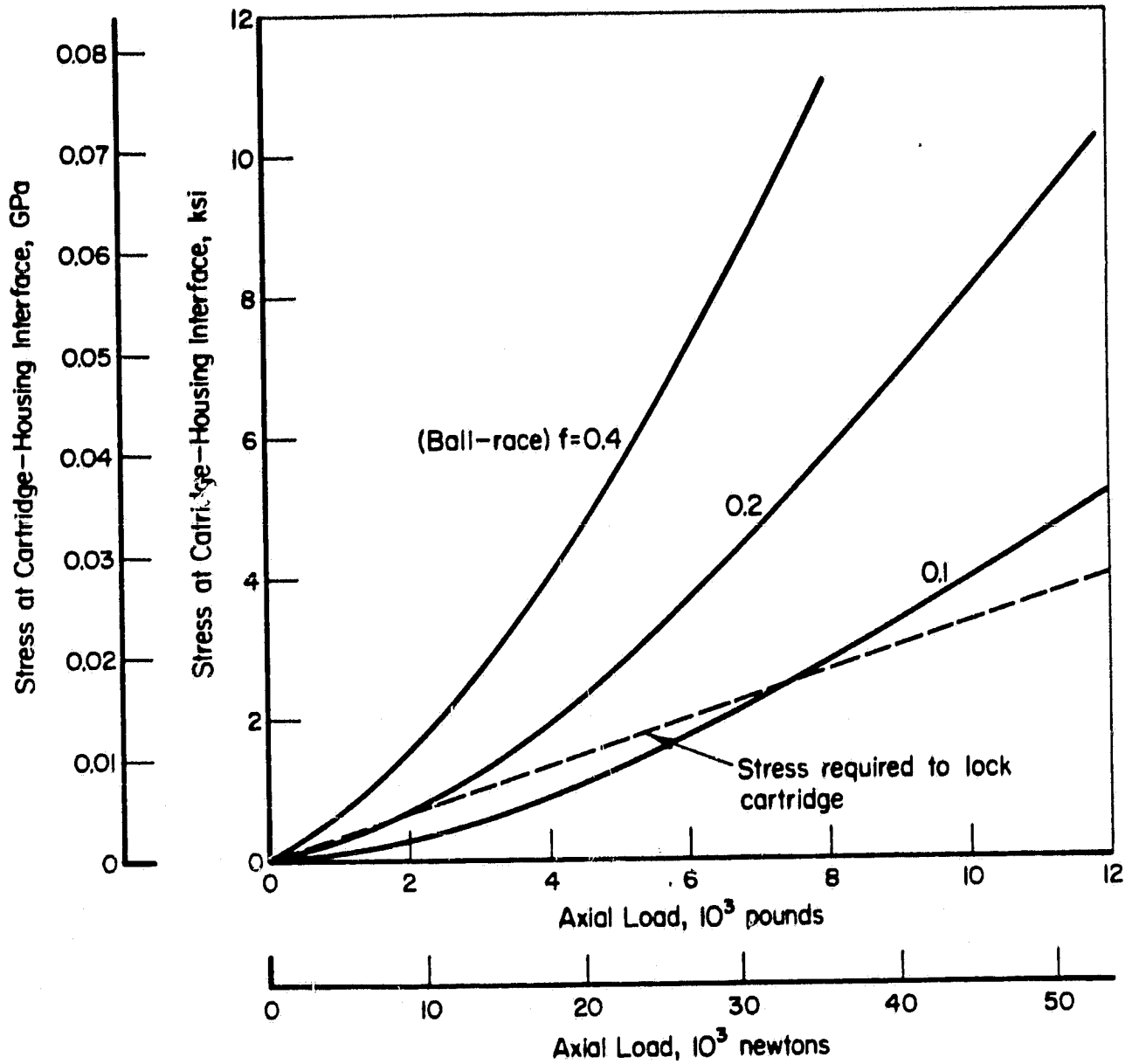


FIGURE 1. EFFECT OF AXIAL LOAD ON CARTRIDGE HOUSING STRESSES

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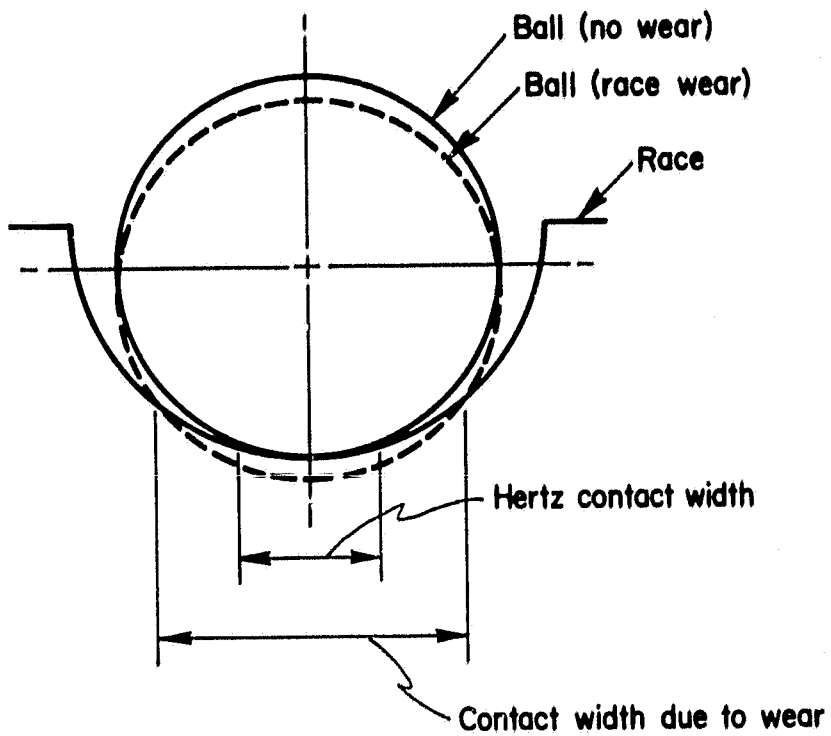


FIGURE 2. ILLUSTRATION OF BALL WEARING INTO RACEWAY

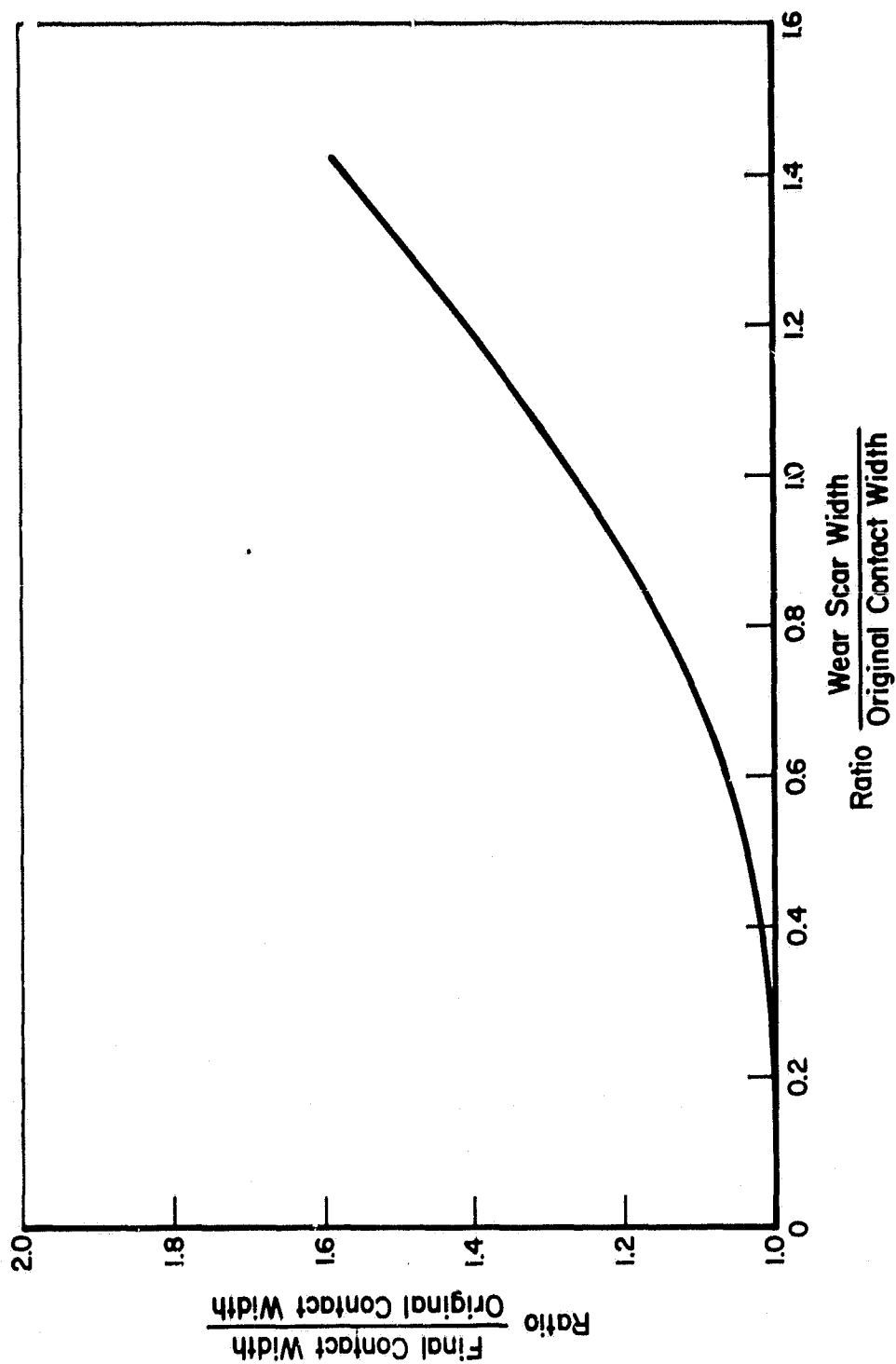


FIGURE 3. EFFECTIVE CONTACT WIDTH AS A FUNCTION OF WEAR SCAR

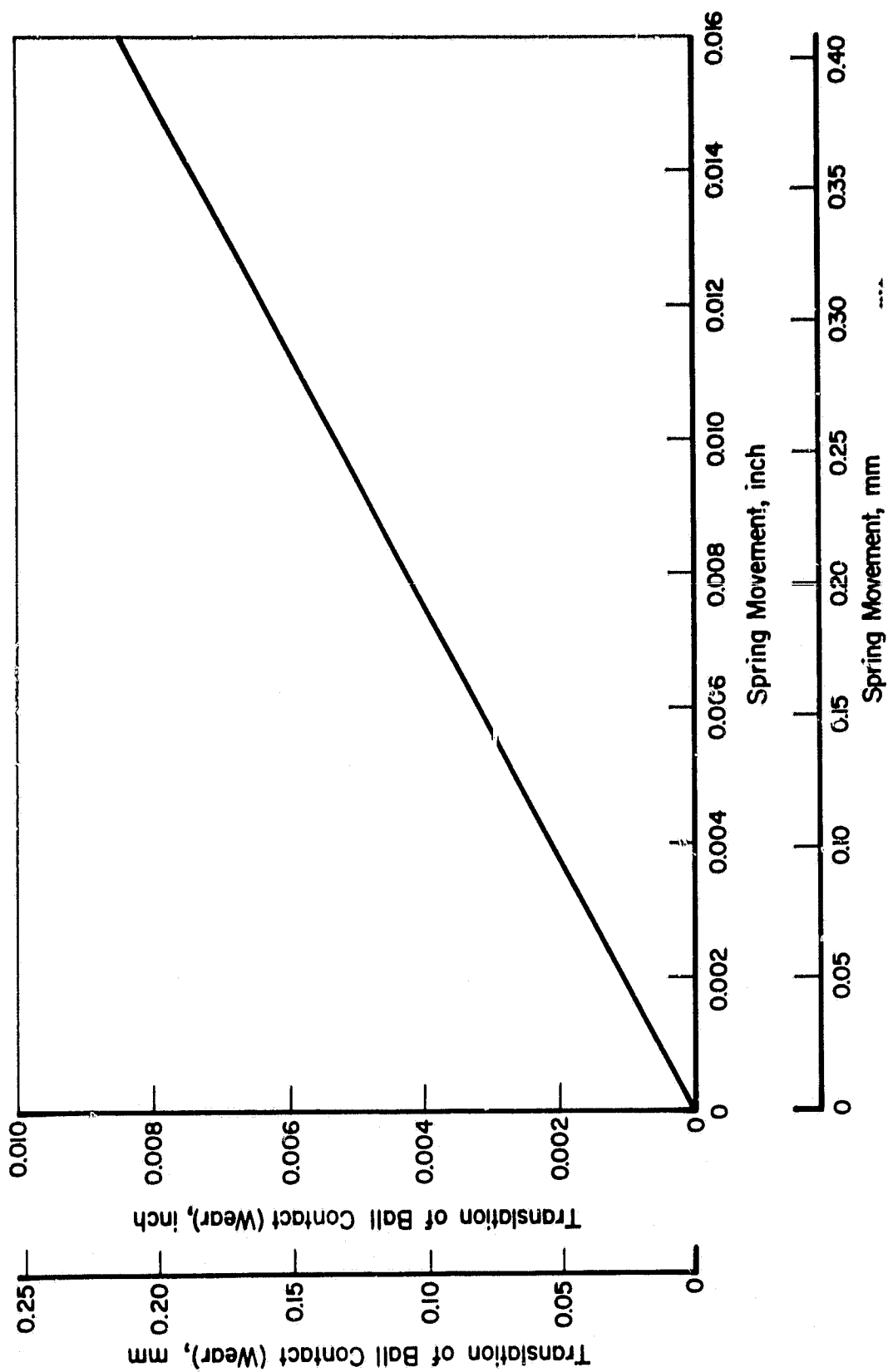


FIGURE 4. EFFECT OF BALL WEAR ON PRELOAD SPRING MOVEMENT

Evaluations were also made of the possible effect of thermal growth of the bearing on wear, as shown in Figure 5. (Figure 5 was based on contact angle and a shaft thermal coefficient of expansion of 10.8 mm/mm/°C.) Essentially, this graph illustrates that a very large (unrealistic) thermal gradient would be required to explain bearing wear of 0.20mm (0.008 inch).

Maximum Stresses as a Function of Load

Bearing life is known to be heavily related to the magnitude of the shear stress beneath the surface of the bearing ball and races and to the depth of those stresses. These stresses and depths are computed as a part of the life calculation in the Battelle bearing dynamics model, BASDAP. As a part of this project, several BASDAP computer runs were made for different loads to compute the fatigue stresses.

Figure 6 presents the maximum shear stress and stress reversals for an SSME 7955 bearing (see Table 1). At a load of 35,580N (8000 pounds) the maximum shear stress is as high as 1.1 GPa (150,000 psi) at the outer race. The maximum shear reversing stress is 0.83 GPa (120,000 psi). It should be noted that the range of shear stress for the reversing stress is twice the maximum values 1.66 GPa (240,000 psi), which is the reason reversing stress is used in fatigue calculations.

The depth of the maximum stress, as computed using BASDAP, is shown in Figure 7. Under loads of 35,580N (8000 pounds), for example, the maximum stress and stress reversal locations are at 0.20mm (0.008 inch) below the surface. In reality, a range of stress depths (rather than a single value) might be more appropriate in defining critical depth for fatigue. That is, the shear stress is very high for at least \pm 30 percent of the maximum value so that the critical depth here could well be .28mm (.011 inch) or even deeper.

Evaluation of Thermal Growth on Spring Load Deflection

The objective of the thermal-wear analyses was to determine the effect of thermally induced shaft diametral growth on spring load deflection and bearing stress. BASDAP was used in these computations. BASDAP

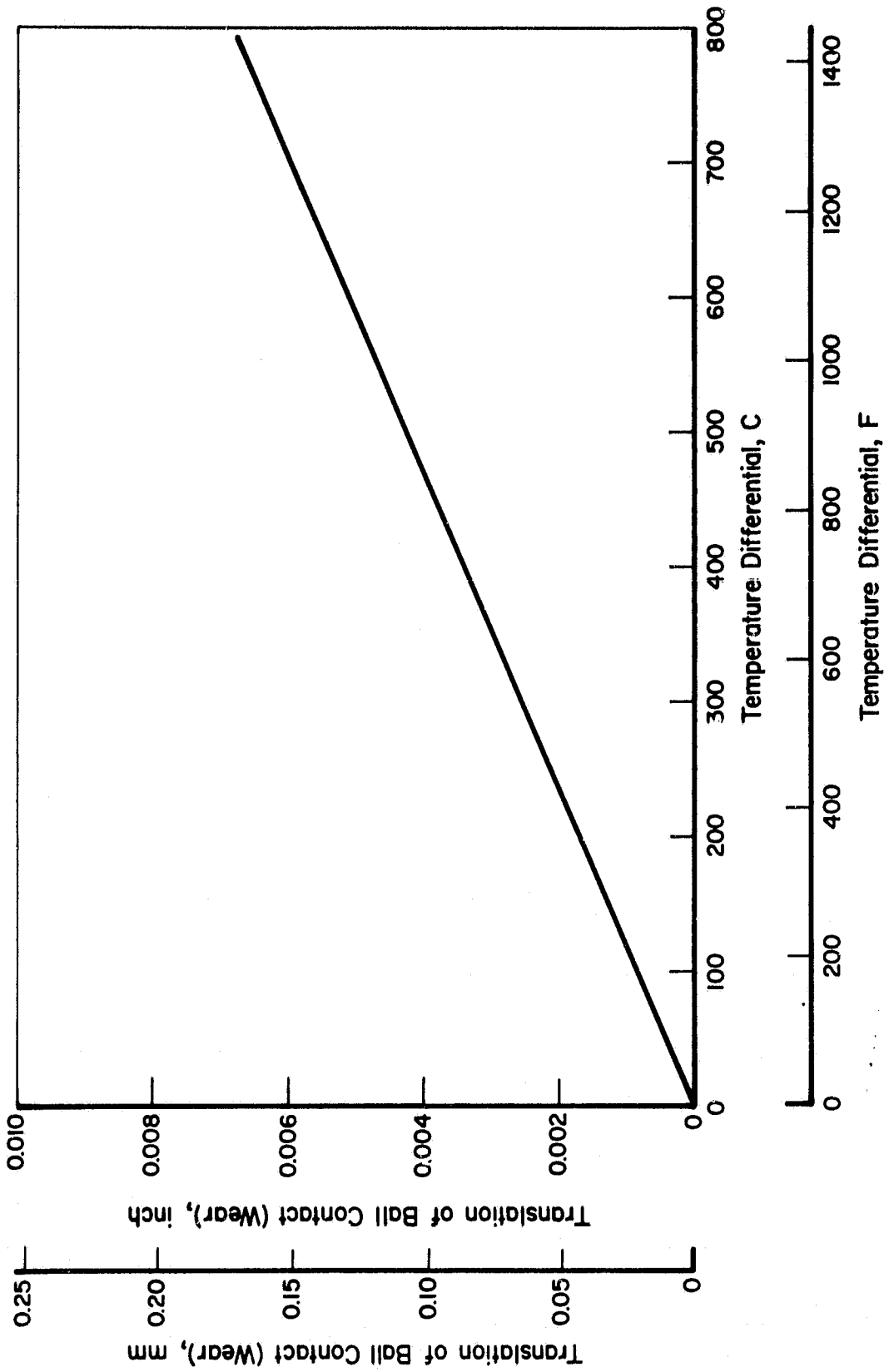


FIGURE 5. TRANSLATION OF CONTACT ZONE (WEAR) DUE TO THERMAL GROWTH OF SHAFT

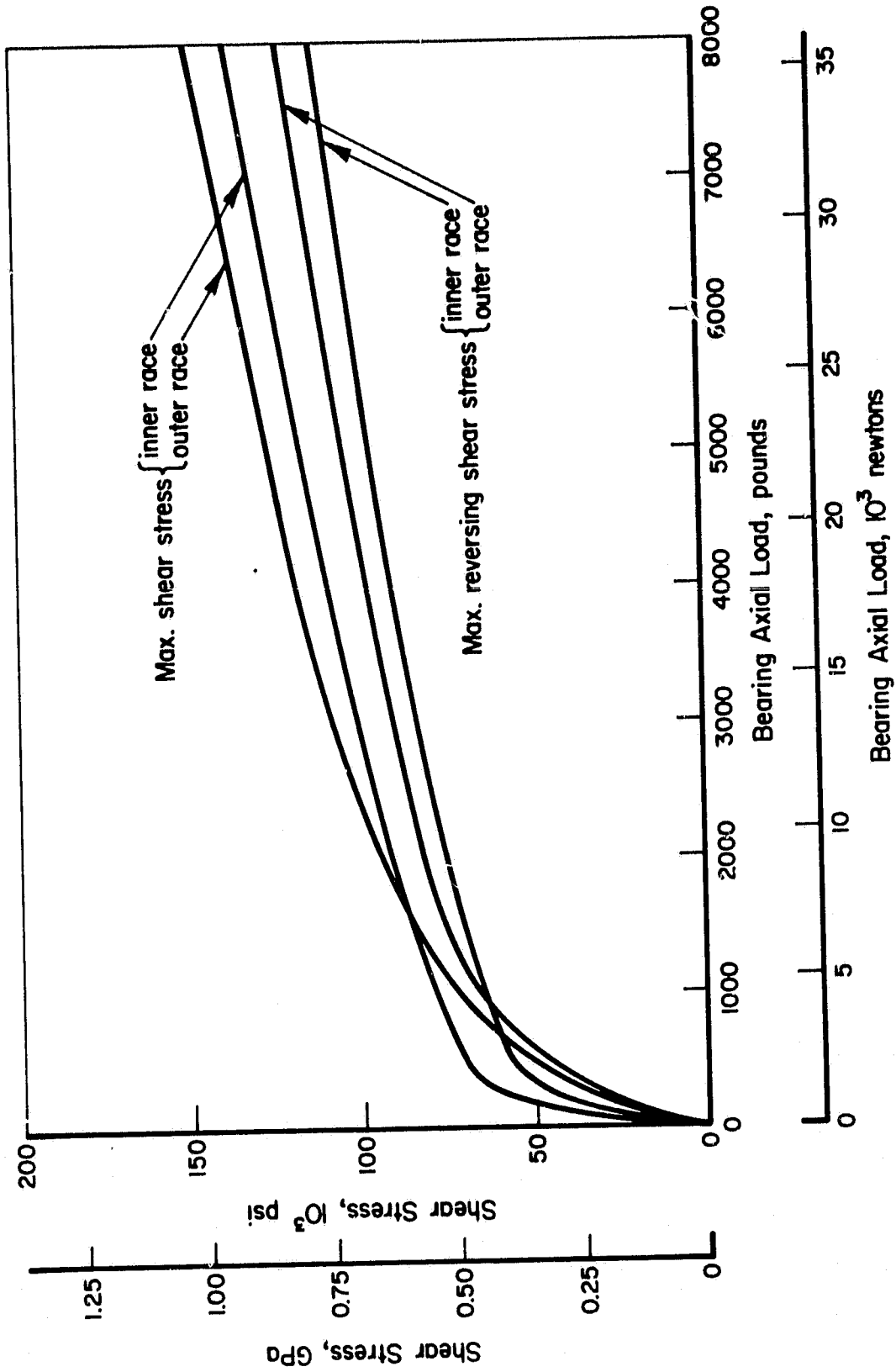


FIGURE 6. MAXIMUM REVERSING STRESS AND SHEAR STRESS IN SSME (7955) BEARING

TABLE 1. INPUT PARAMETERS FOR BEARING 7955

Parameter	Symbol	Numerical Value
Pitch Radius	R_p	40.5mm (1.595 inches)
Ball Radius	R_B	6.35mm (0.25 inch)
Number of Balls	NB	13
Inner Race Curvature	f_2	0.53
Outer Race Curvature	f_1	0.53
Design Contact Angle	β	15°, 25°
Thermal Conductivity	K	29.2 N/sec-C (3.65 lbs/sec-F)
Thermal Expansivity	α	12.6 x 10 ⁻⁶ cm/cm/C (7x10 ⁻⁶ in/in/F)
Operation Speed	Ω	30,000 rpm

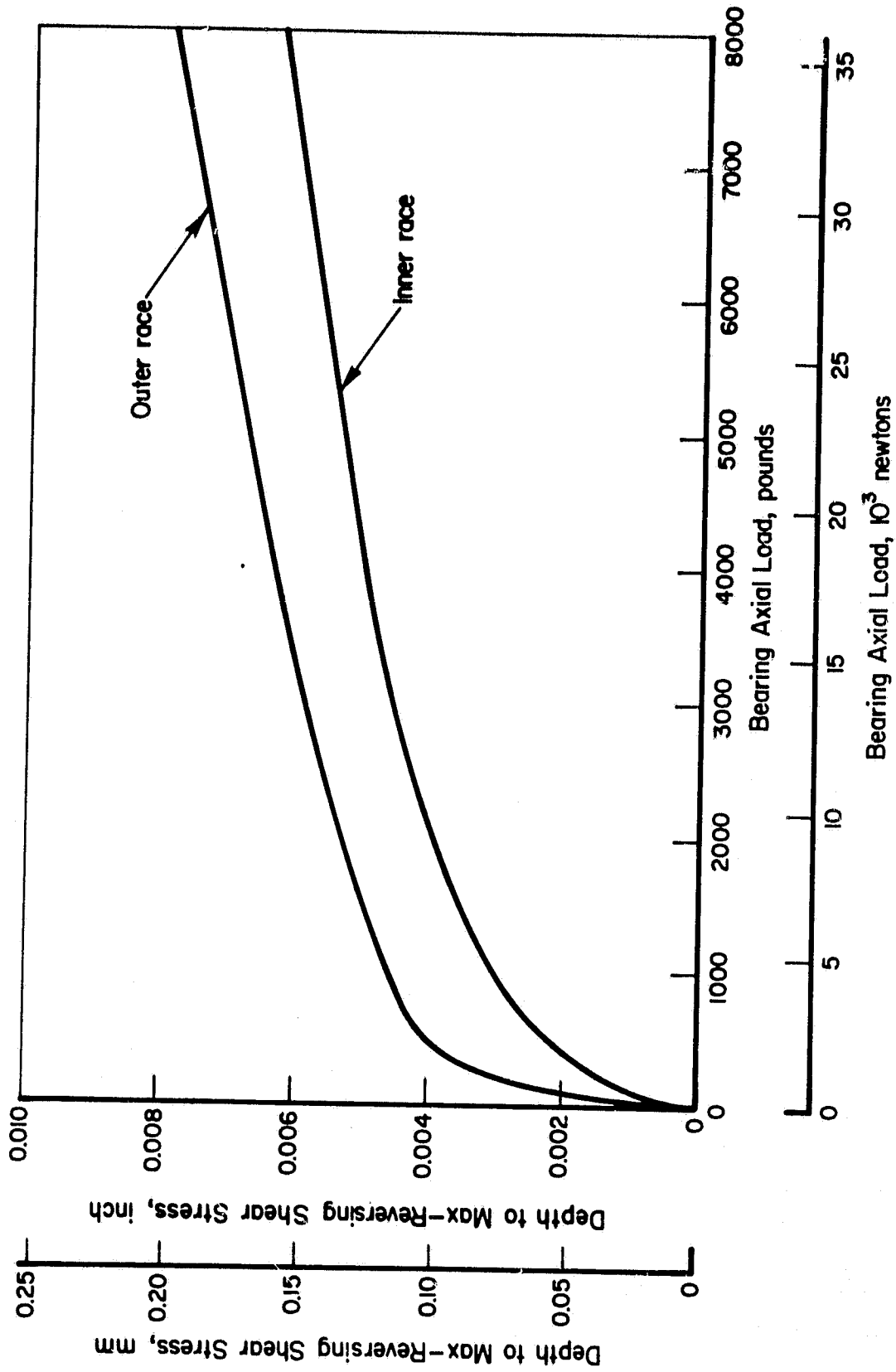


FIGURE 7. DEPTH OF MAXIMUM SHEAR REVERSALS FOR SSME (7955) BEARING

normally computes bearing stresses and deflection for fixed preloads. However, the preload spring deflection and preload are interrelated. Therefore, modifications to the BASDAP model were required in the force balance subroutine.

Normally a bearing deflection condition is assumed and the load on the bearing is calculated. This computed load is compared with the known load and the deflection adjusted accordingly. An iterative scheme is used to find the correct deflection corresponding to a given load. In the modified program, an additional iterative scheme is used to allow the bearing deflection to alter the preload (by the preload spring).

Figure 8 shows the computed effect of temperature differential on preload spring motion. A temperature of 100C (210F) would cause a deflection of about 0.08mm (.003 inch). Figure 9 shows how the preload is altered by the spring movement due to temperature. At a 100C (210F) temperature differential, only a minor change in preload occurs. However, as the springs bottom out, temperatures cause a marked change in load. This load increases due to loss of preload spring movement can be a cause of bearing distress.

Calculating Units

Since the bearing drawings and all input data provided by NASA were in English units, all calculations were performed in English units. Therefore, the SI units presented in this report were converted from English units.

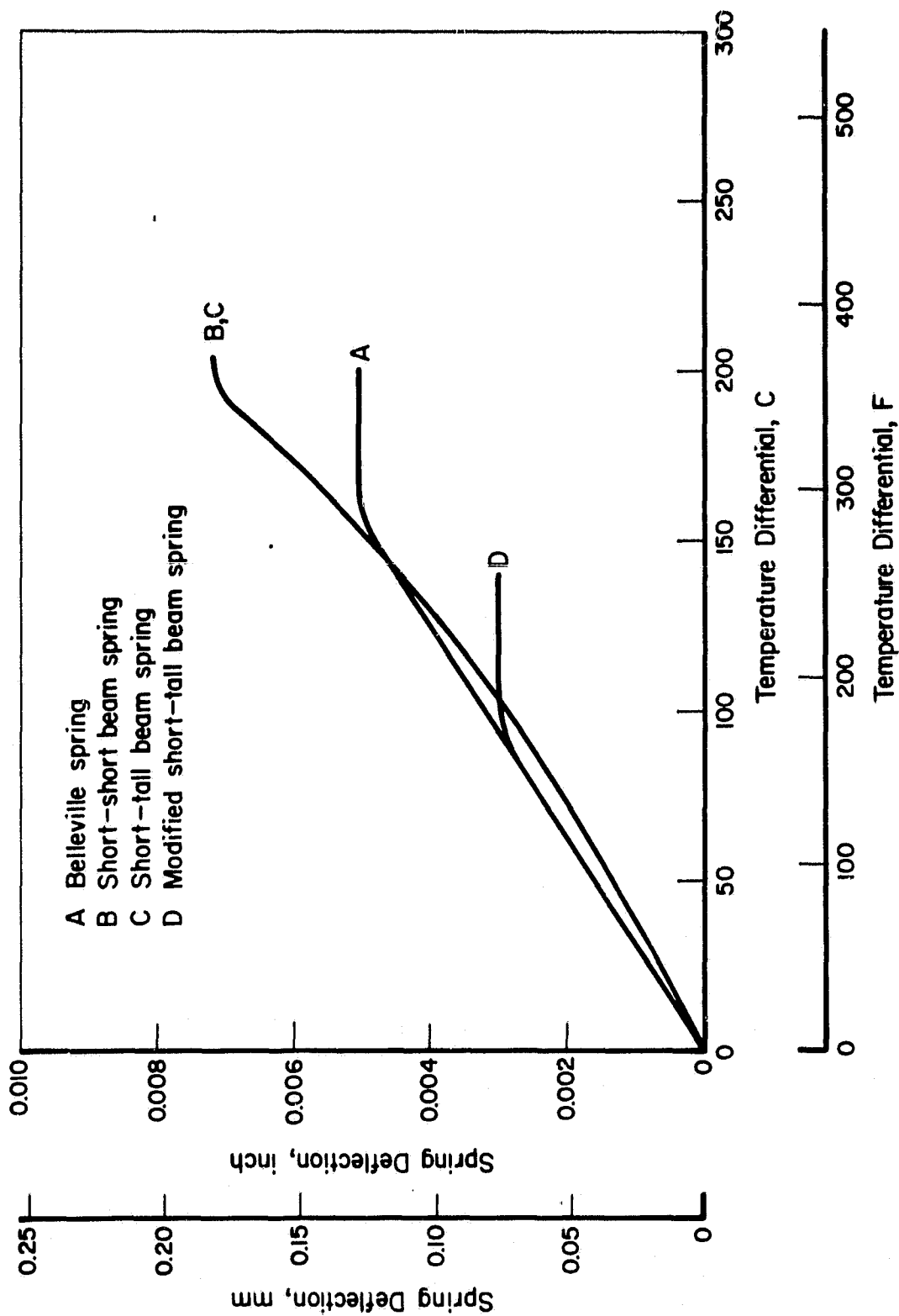


FIGURE 8. EFFECT OF INNER-OUTER TEMPERATURE DIFFERENTIAL ON AXIAL SPRING DEFLECTION

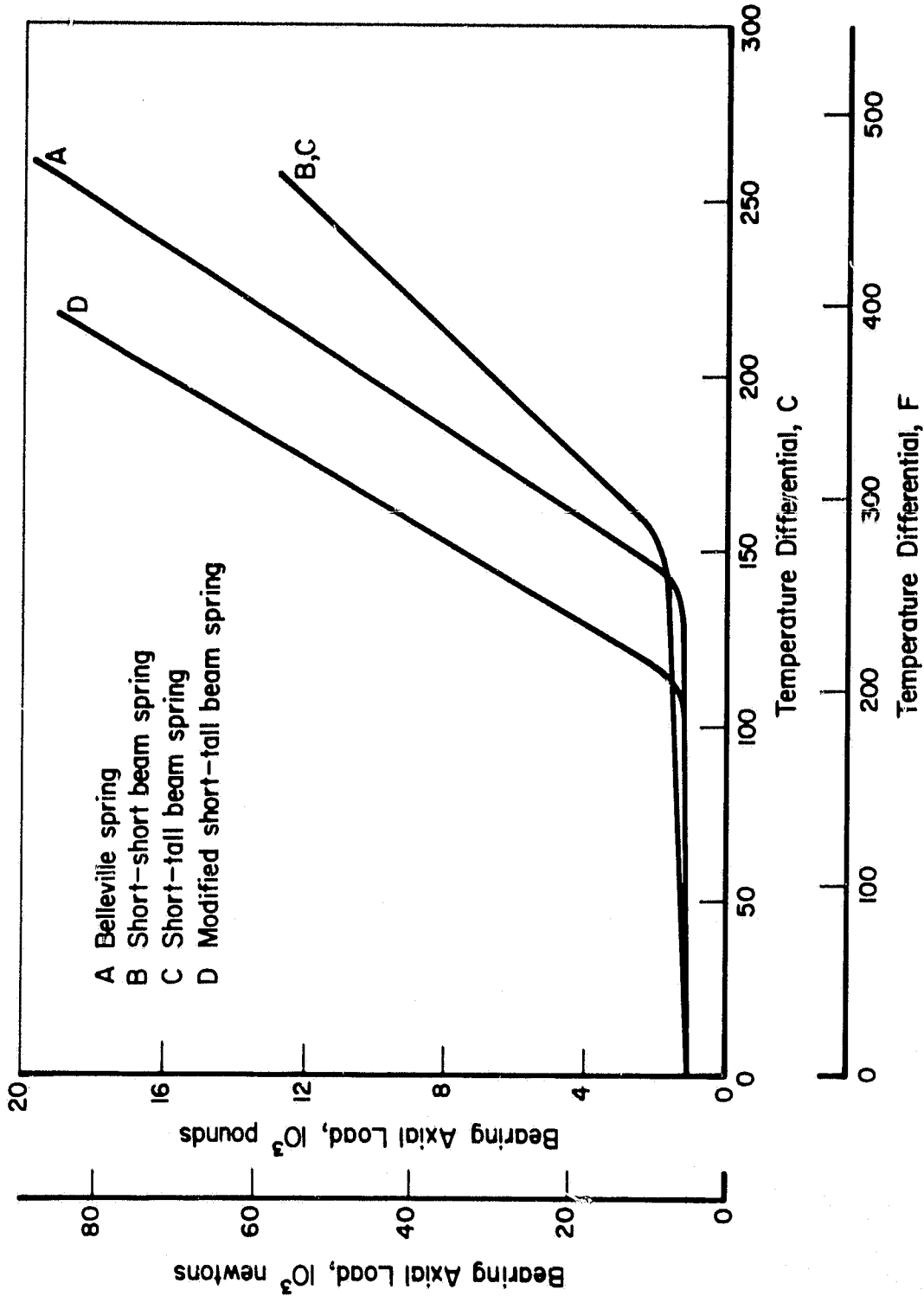


FIGURE 9. EFFECT OF INNER-OUTER RACE TEMPERATURE DIFFERENTIAL ON AXIAL LOADING

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- (3) Smith, G., and Dolan, F., Viewgraphs from April, 1982, presentation.
- (4) Kannel, J. W., and Merriman, T., "SSME Turbopump Bearing Analytical Study", to NASA-GCMSFC, August 20, 1980.